Influence of design parameter variability on the dynamic behaviour of honeycomb sandwich panels

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Abstract

Honeycomb sandwich panels are layered structures that consist of at least five layers: two thin face sheets are bonded to a thick honeycomb core. Because of the wide range of panel parameters, numerical modelling is needed to provide insight into the structural characteristics of a particular panel. In this paper the effect of design parameter variations on the dynamic behaviour of honeycomb sandwich panels, in particular of thermoplastic Monopan panels, is studied. In the first section the specific structure of Monopan honeycomb panels is illustrated. The different design parameters of panels of this type are outlined. The second section deals with the estimation and experimental identification of the different design parameters. Several sandwich parameters are found to exhibit a significant amount of scatter. The third section deals with the numerical modelling of the panels. Estimated and measured average parameter values are used to obtain a good initial FE model. The fourth section covers the experimental validation of the simulation results for a number of test panels. Measurement uncertainty is discussed in detail. The fifth section presents the model updating procedure. Sensitivity analysis is used, and global and local model updating is discussed.

1 Introduction

Honeycomb sandwich panels consist of a thick honeycomb core that is bonded to thin face sheets. The structure of a typical panel is shown in fig. 1. The coordinate system is used throughout this text, although the axes are often indicated with numbers 1 to 3. The honeycomb panels Monopan® panels. Panels of this type have a cylindrical cell honeycomb core made of polypropylene (PP). The core is welded to a Twintex skin by means of a welding foil. The Twintex skin consists of a symmetric glass fibre woven fabric with a polypropylene matrix and a theoretical thickness of approximately 0.7 mm. To smoothen the outer surfaces of the panel, a polypropylene finishing foil is welded there.

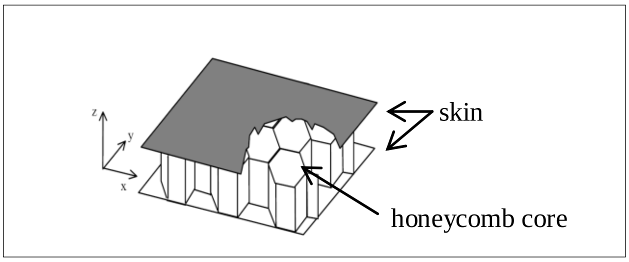

Figure 1: a typical honeycomb sandwich panel

Monopan panels of different sizes (length/width and thickness) are used. A first set of 7 panels have an overall thickness of 15 mm and in-plane dimensions 594 x 420 mm (A2 size). A second set of 7 are 25 mm thick and are also A2 size. Panels of the third set of 7 have a thickness of 25 mm and larger dimensions: 2500 x 1200 mm. All panels have the same structure, only the core height is different.

The elastic mechanical properties of a typical honeycomb core are described and analytically calculated by Gibson & Ashby [1]. They propose formulas for calculation of the in-plane and out-of-plane elastic moduli and Poisson ratios of the core.

As honeycomb sandwich panels become more and more important as structural parts in the automotive and aerospace industry, the need increases for accurate modelling of the dynamic behaviour of such panels. Accurate models require knowledge of the different design parameters that determine the dynamic behaviour, which in this case is described by natural frequencies and mode shapes of panels with free-free boundary conditions.

A large portion of the literature on the dynamics of sandwich panels is related to conventional foam-core structures. Some work has also been carried out on honeycomb panels. Nilsson & Nilsson [2] tried to analytically predict natural frequencies of a honeycomb sandwich plate with free-free boundary conditions using Blevins [3] formula in which areal mass and equivalent bending stiffness are frequency dependent; this to include visco elastic material behaviour. Another, more practical way to predict natural frequencies and mode shapes of a honeycomb panel is by means of finite element analysis. In the past years, different new approaches have been developed which incorporate high order shear deformation of the core. Work in this area has been carried out by Topdar [4] and Qunli Liu [5][6][7]. The latter stated that the shear moduli of the core are important factors in the determination of the natural frequencies and the sequence of mode shapes, especially at high frequencies. At low frequencies natural frequencies are mostly determined by the bending stiffness of the panel.

The present analysis identifies parameter variability, with the definition of variability as given in [8]. Design parameter variability of Monopan panels is studied, along with its influence on the dynamic behaviour of such panels subjected to free-free boundary conditions. For 1D laminated structures an approach to this problem is addressed in [9]. This work mainly focuses on the inverse problem, identifying material properties for layered materials by experimental identification of the vibration behaviour. A recent approach in dealing with design parameter uncertainty is describing modal parameters as random fields. The random field theory is extensively described by Ghanem [10]. Schuëller [11], Soize [12] and Desceliers [13] recently have adopted this theory for inverse problems and for cases where limited experimental data is available.

2 Design parameter determination

The Monopan honeycomb panel structure shown in fig. 2 has a high number of design variables. It is therefore difficult to accurately predict their dynamic behaviour analytically or by means of a simple finite element model. In addition, some design parameters are very difficult of even impossible to determine experimentally.

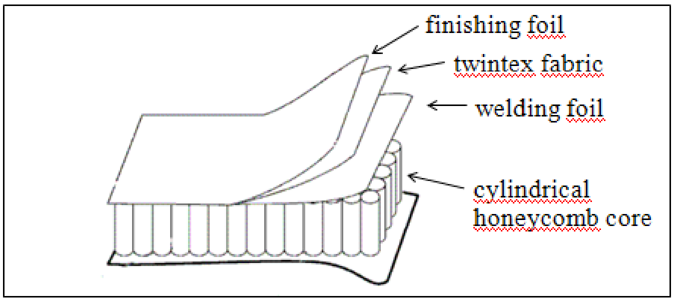

Figure 2: different sandwich layers of a Monopan honeycomb panel

geometric parameter		unit	material parameter		unit
core material elastic modulus	E_{cm}	MPa	overall panel width	w	mm
core material poisson ratio	μ_{cm}	-	overall panel length	l	mm
core material shear modulus	G_{cm}	MPa	skin thickness	t_s	mm
core material mass density	ρ_{cm}	kg/m³	core thickness	t_c	mm
welding foil elastic modulus	E_{wf}	MPa	core cell inner diameter	d_{ci}	mm
welding foil poisson ratio	μ_{wf}	-	core cell wall thickness	t_{cw}	mm
welding foil shear modulus	G_{wf}	MPa	finishing foil thickness	t_{ff}	mm
welding foil mass density	ρ_{wf}	kg/m³	welding foil thickness	t_{wf}	mm
finishing foil elastic modulus	E_{ff}	MPa	parameter description	symbol	mm
finishing foil poisson ratio	μ_{ff}	-	overall panel width	w	mm
finishing foil shear modulus	G_{ff}	MPa	overall panel length	l	mm
finishing foil mass density	ρ_{ff}	kg/m³			
Twintex matrix elastic modulus	E_{tm}	MPa			
Twintex matrix poisson ratio	μ_{tm}	-			
Twintex matrix shear modulus	G_{tm}	MPa			
Twintex matrix mass density	ρ_{tm}	kg/m³			
Twintex fibre elastic modulus	E_{tf}	MPa			
Twintex fibre poisson ratio	μ_{tf}	-			
Twintex fibre shear modulus	G_{tf}	MPa			
Twintex fibre mass density	ρ_{tf}	kg/m³			
skin fibre volume fraction	v_{tf}	%			

Table 1: List of design parameters.

The panel face sheet, for instance, is in itself a complicated structure consisting of three layers (see figure 2) that are not easy to distinguish visually. Scatter on experimentally measured thicknesses of these three layers is high, because of variability that is physically inherent to the production process. In this study these thicknesses are determined by analysis of microscopic images and by micrometer measurements. For example, the thickness of the PP filled Twintex fabric is determined experimentally. The histogram is shown in figure 3. A mean value of 0.699 mm is obtained from the measurements but the histogram clearly shows that there is a large variability. Other design parameters can be measured more directly, which result in less variability. Figure 4 shows the histogram of the inner diameter of the cylindrical honeycomb core cells. This was experimentally determined with a 3D CNC measurement bench; 240 cells were measured, each determined diameter resulting from the least squares best fit through 10 measurement points on the cell inner circumference. For this parameter a mean value of 7.84 mm is found from the measurement. Figure 4 clearly shows that the relative range on this parameter is much more narrow than that of the face sheet thickness. At this stage of the research only global parameter variability is studied; spatial variability of the different design parameters has not yet been addressed. As the first objective of this study was the determination of a mean computational FE model to predict the vibrational behaviour of the panels the focus of this work was on the determination of mean parameter values, together with their respective probability distributions Some design variables cannot be determined directly through experiments as for example mass density and fibre volume fraction. To obtain reliable estimated mean values for those parameters a combination of experiments and estimations is used to calculate the total panel mass, which in itself is a measured quantity. In the calculation of panel mass all constituting sandwich layers are considered to have a uniform thickness. The panel dimensions and the layer thicknesses are measured quantities and mass properties are estimated. However, from a single mass calculation it is not possible to obtain correctly estimated values for the mass variables. A Monte Carlo (MC) scheme is set up to overcome this. For convenience the measured quantities are considered to be normally distributed around their measured mean value and in an interval that equals s the measured probability interval of each parameter.

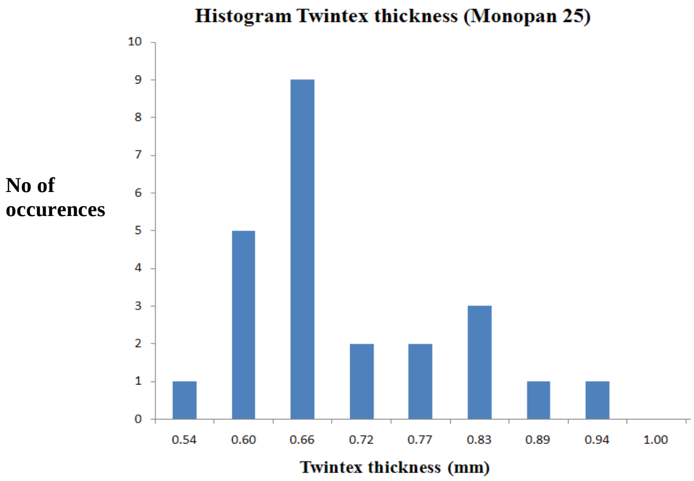

Figure 3. Histogram of measured Twintex thickness

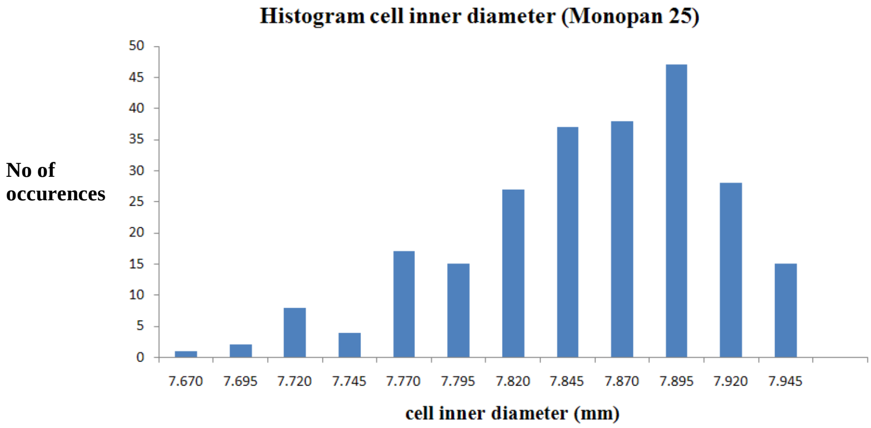

Figure 4. Histogram of measured cell inner diameter

The estimated variables are also considered to be normally distributed within an specific interval which is not known a priori. The random variables are the measured quantities discussed above, considered in their measured interval. (During the same iteration step a optimization is performed to minimize the difference between the (random) measured panel mass and the calculated panel mass. This optimization yields a set of values for the estimated parameters. Table 2 gives an overview of measured and estimated mean parameter values as input for the algorithm in case of a Monopan panel with dimensions 595 x 420 x 15 mm. The width of each (symmetric) probability interval is also given.

param.	**m (kg)**	**w (mm**	**l (mm)**	**t (mm)**	**t_s (mm)**	**t_{tw} (mm)**	**ρ_{cm} (kg/m³)**	**ρ_{tf} (kg/m³)**	**ρ_p (kg/m³)**	**v_{tf} (%)**
mean	1.005	419.9	594.7	15.15	1.2006	0.6997	68	2550	1150	20
Interval (%)	1	0.5	0.5	1	5	5	5	2	2	10

Table 2: mean measured and estimated panel mass parameters.

This Monte Carlo algorithm is carried out 50000 times. As an example figure 5 shows the histogram for the estimated glass fibre mass density ρ_{tf} of the Twintex fabric.

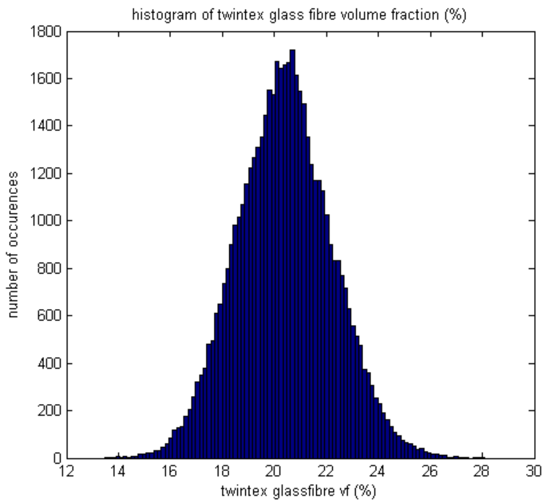

Figure 5: histogram of Twintex glass fibre volume fraction

Table 2 gives the mean and standard deviation values for the four estimated mass governing parameters.

parameter	**ρ_{cm} (kg/m³)**	**ρ_{tf} (kg/m³)**	**ρ_p (kg/m³)**	**v_{tf} (%)**
mean	68.254	2588	1141	20.4
standard deviation	0.97	20.17	15.96	0.018

Table 2: mean and standard deviation for estimated panel mass parameters.

All histograms have a nice shape and standard deviations of the different estimated parameters are small in comparison to the corresponding mean value. In addition the mean value of all calculated panel masses is 1.0039 kg which is very close to the mean experimentally determined value. The measured mean parameter values from table 1 and the estimated ones from table 2 are used to construct a FE model of the panel.

3 Numerical modelling

A physically realistic FE model of a honeycomb sandwich panel has a high number of degrees of freedom. Even for a small size panel this approach yields a very high number of finite elements and nodes.

A more suitable method of modelling a whole honeycomb sandwich panel is to use some degree of homogenisation in order to reduce the number of govering parameters. A full homogenisation of the panel is computationally attractive but this approach yields unsatisfactory results as the differences between calculated and measured natural frequencies increase dramatically with increasing frequency because the shear deformation of the core is not correctly taken into account. The so called SVS–concept ('Shell – Volume – Shell') is a very good compromise between accuracy and computational efficiency. This method homogenises the honeycomb core and the skin. As the core height is about the same order as the total panel thickness, the core itself is meshed with volume elements. The skin thickness on the other hand, is much smaller than the total panel thickness, so shell elements can be used here. In addition bending is the most significant mode of deformation of the panel.

The honeycomb core has a periodic cylindrical pattern. Because of the spatial symmetry it is homogenised as an orthotropic material. To characterise the elastic behaviour of an orthotropic material, 9 independent constants have to be determined. For the honeycomb core of the Monopan panels, these 9 constants are determined by modelling a unit cell of honeycomb core material and by loading it with tension and shear along the 3 planes of symmetry. The unit cell itself is modelled using shell elements and experimentally determined mean parameter values for e.g. core height, cell inner diameter, cell wall thickness and mass density are used. An example is shown in figure 6. Note the axes used.

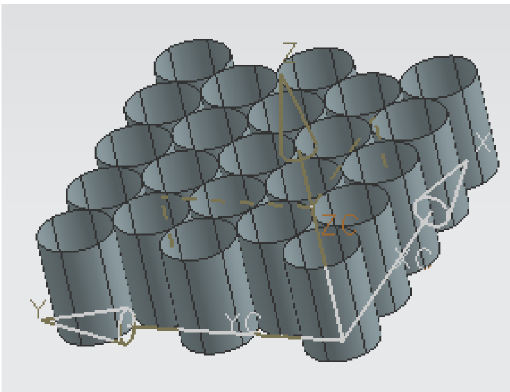

Figure 6: Unit cell of Monopan cylindrical honeycomb material

To validate the results of the core homogenisation, two sets of experiments are carried out. In a first test the elastic modulus along the Z – axis E_3 is determined experimentally by carrying out a set of 50 compression tests. These tests yield 124.1 MPa as a mean value for the homogenised through-the-thickness modulus E_3, while the calculated value from the model homogenisation is 130 MPa.

In a second test the out-of-plane shear modulus G_{13} is determined. This is done by carrying out 3–point bending tests on Monopan beam samples. Equation (1) gives the total deflection of a beam as the sum of the contributions of bending deformation and shear deflection [14].

$$\frac{d}{FL} = \frac{L^2}{48D} + \frac{1}{4AG} \tag{1}$$

In equation 1, d is the deflection under the centre load F, L is the span length, D is the pure bending stiffness and G is the core shear modulus. A is $\frac{b(t_c + t_s)^2}{t_c}$ where b is the width of the beam and t_c and t_s are respectively the core height and the skin thickness. Two sets of 20 bending tests are carried out for span lengths 300 and 500 mm, giving a mean value of 61 MPa for G. The calculated value for G_{13} is 64 MPa.

For the tests on the two different span lengths little difference between calculation and measurement occurs, so the homogenisation of the honeycomb material is justified.

The skin properties are determined using a similar procedure. According to Ishai [15], materials reinforced with a woven fabric can be approximated by a laminate structure. If the woven fabric is symmetric in weft and warp direction, as in the Twintex case, the elastic behaviour of this laminate can be modelled as an orthotropic material. The two other layers in the skin, the welding and finishing foils, are treated as isotropic materials. Eventually the whole skin is homogenised as an orthotropic material. To model the skin, again 9 independent elastic constants have to be determined. As the skin is relatively thin in comparison to the thickness of the whole panel and since the skins are located on the outer sides of the sandwich panel, only the in – plane (xy - plane) elastic properties of the skin are important for the elastic behaviour of the whole panel, especially the elastic moduli E_{s1} , E_{s2} and the shear modulus G_{12}. Since the Twintex fabric is symmetric, moduli in directions 1 and 2 are expected to be equal. To determine this modulus of elasticity a series of 30 tensile tests have been carried out on beam samples (25 mm wide and 200 mm long). The experimental results show a large amount of scatter. This is partially due to local effects near the clamped sides of the specimens. A mean value of 10.8 GPa is found while the manufacturer of the Twintex reinforced Polypropylene specifies a value of 14 GPa.

The final SVS – finite element model thus consists of 3 orthotropic layers. This model still has a rather large number of design variables: 18 elastic constants, 2 mass densities and 4 dimensions (panel dimensions and layer thicknesses). A sensitivity analysis is helpful to find out which of these 24 model parameters are dominant for the dynamic behaviour of the honeycomb panels.

The minimum number of elements in the FE model is determined by checking the convergence of the solutions when increasing the number of elements in the FE model For the A2 size Monopan 15 panel for instance, convergence to a steady solution is reached when 80 x 48 x 9 elements are used. Figure 7 shows the convergence curves for modes 1, 5 and 10.

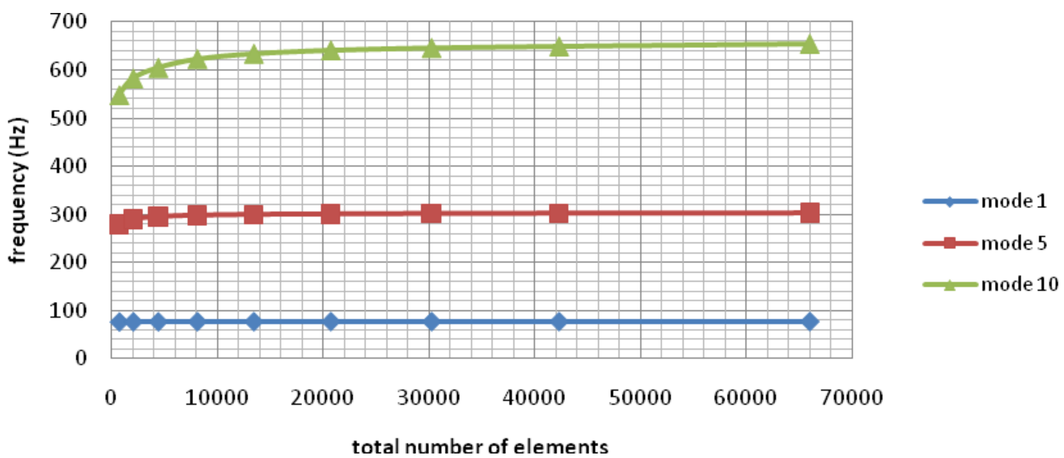

Figure 7: convergence curves for SVS FE model of a Monopan panel (595x420x15 mm)

As mentioned earlier, the finite element model is used to calculate natural frequencies and mode shapes of panels with free-free boundary conditions. These boundary conditions are simple to model and also relatively easy to realise in experiments. For each of the three types of panels 20 modes are calculated. The results are discussed when compared with the experimental results in section 4. At this stage of the research damping is not yet studied.

4 Experimental validation

In this study natural frequencies and mode shapes of Monopan panels with free-free boundary conditions are determined experimentally. The boundary conditions are achieved by suspending the panels by elastic springs. As mentioned earlier, 3 sets of 7 panels each were used for the measurements. At this stage of the research the goal of the measurements is to have an idea of the scatter on the natural frequencies of a set of virtually identical panels. In addition we are interested in repeatability and accuracy of the vibration measurements. Honeycomb panels are light-weight structures. They have a high overall bending stiffness but rather poor local stiffness. This is why caution has to be made when panels of this kind are locally excited to obtain frequency response functions (frf). In this context a fully contactless measurement method was applied as a test case. The panel is excited by a loudspeaker and vibration is monitored with a laser vibrometer. Although this technique has the advantage that no local deformation is applied and no extra mass is added to the structure, it yields incorrect frequency response functions due to the input force that is not known exactly. In turn this yields locally perturbated mode shapes. So this method is only applicable for determining resonance frequencies. The measurement approach that is discussed further is the classic hammer excitation method. When a number of test measurements are performed on a Monopan panel with dimensions 595x420x15 mm some interesting issues about measurement variability can be discussed. The repeatability of the experimental determination of resonance frequencies is very high; a series of measurements yields that variability of those frequencies is as small as the accuracy set in the FFT analyzer of the data acquisition system used. However, frf amplitudes at resonance peaks are found to exhibit a much larger range of variability. Table 3 shows the measured frf amplitudes at the first four resonances. All frequency response functions have the same excitation and measurement point locations.

#	A 1 (m/Ns^2)	A 2 (m/Ns^2)	A 3 (m/Ns^2)	A 4 (m/Ns^2)
1	28.9	18	20.3	3.85
2	30.8	19.6	21.8	3.99
3	30.5	19	21	3.94
4	28.9	18.9	20.8	3.6
5	29.3	18.5	20.8	3.93
6	28.5	18.3	20.1	3.59
7	28.5	18.3	20.1	3.59
8	28.6	18.5	20.2	3.56
9	29.5	18	20.6	4.07
10	28.7	18	20.3	3.77
11	28.7	18.4	20.5	3.66
12	29.3	18.2	20.5	3.92
min	28.5	18	20.1	3.56
max	30.8	19.6	21.8	4.07
mean	29.183	18.475	20.583	3.789
scatter (%)	7.881	8.660	8.259	13.459

Table 3: overview of measured frf amplitudes for the first four modes (Monopan 15, panel 7).

The large scatter on the measured frf amplitudes at resonance may have several causes. A first one is that the accelerometers and the excitation hammer used could result in erroneous measurements. However, all of these were calibrated before the experiments. In addition, calibration of hammer and accelerometers was exercised 25 times to study repeatability. Hereby the excitation hammer was calibrated using a freely suspended steel mass and a reference accelerometer. By simply hitting and monitoring the force and acceleration peak amplitudes the mass is calculated by Newton's law. During these measurements a small variability of about 0.41 % on the derived mass resulted from measurements. It is important to mention that during these measurements the coherence between the excitation signal and the acceleration signal was always excellent. It can be stated that frf measurement variability is not resulting from transducer inaccuracy. There are a number of other possible causes for the experienced variability on frf amplitudes. First of all there is some variability on the excitation point location. Although the excitation points are marked on the honeycomb panels, it is never possible to hit the panel exactly at the excitation location. To study the effect of an excitation point location variability on a measured frequency response function, some simulations were made to illustrate this effect. Figure 8 shows a few frequency response functions. The first one is the reference frf, with the second one the excitation point is moved 31 mm in the x – direction (horizontally) and with the third one the excitation point is moved 25 mm in the y direction (vertically). The simulation of a big panel with dimensions: 2500x1200x25 mm is made.

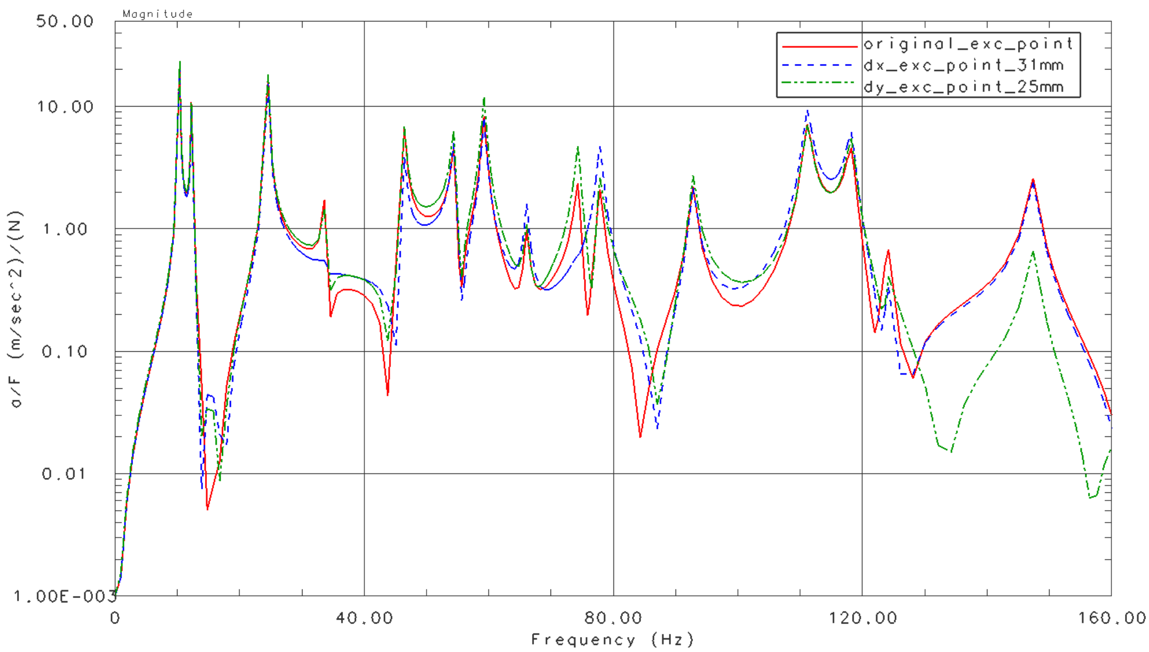

Figure 8: alteration of the excitation point location (Monopan 2500x1200x25 mm)

When the frf amplitudes at the third mode at approximately 23 Hz are compared, the horizontal perturbation (31 mm or 1.24 % of panel length) causes a 10.4 % change in frf amplitude. The vertical perturbation (25 mm or 2.08 % of panel height.) causes a change of 4.36 %. A realistic typical error on the excitation point location was easily determined experimentally by painting the tip of the excitation hammer. A set of 25 hits, aimed at the same excitation point, yielded a circular domain with a radius of approximately 4 mm. For the mode studied, a worst case error of 4 mm on the excitation point location could thus result in a calculated change in the frf amplitude of about 1.3 %.

A second possible cause of measurement variability is the non-linear behaviour of the panel structure. When hitting a panel, local deformations can occur, resulting in incorrect frequency response functions. Local deformations can occur mainly at two levels. First of all there can be a local deformation of the sandwich skin due to the hammer impact. This deformation is restricted by the bounds of one cell, being the cell wall. Secondly a deformation of the panel structure can occur in a wide zone around the impact location. To study the effect of the first type of local deformation, measurements were carried out using a simple test set-up shown in figure 9. A piece of Monopan panel material is bonded to the steel mass. The mass is freely suspended on light foam material. As with the impulse hammer calibration described above, the amplitude peak values of the excitation force and mass acceleration were registered and the mass was calculated. A series of 30 tests were carried out. The calculated mass experiences a scatter of 3.57 % around the mean value. A resulting 3.1 % scatter on frf resonance amplitudes can thus be attributed to local deformations of the panel structure. One would expect that the effect of local deformation increases with excitation force. However, in the typical excitation force range from approximately 40 to 110 N, no systematic non-linear relation was found between the excitation force and measured mass acceleration. Non-linearities become strongly evident when the excitation force is higher than 140 N. This is probably due to local buckling of the cell walls.

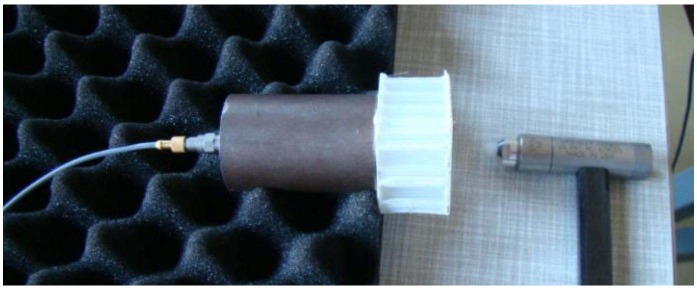

Figure 9: experimental determination of the effect of local deformation due to impact

As a summary, a measurement variability of approximately 5 % results from variability on measurement system performance, the effect of small local deformations of the structure, leading to erroneous frequency response functions and the variability on the excitation point location. It has to be mentioned that the effect of the latter depends strongly with the position of the excitation point. The resulting 3 to 5 % of the measurement variability may then be the result of local deformations of the panel structure in the zone surrounding the excitation point. This is illustrated by carrying out a set of measurements on a panel with dimensions 2500x1200x25 mm. With the same mode shapes as in the case of the small panel (see table 3) an frf amplitude variability ranging from 15 to 20 % is recorded. A sufficient number of averages should thus be taken for each measurement. In addition, in case of the panels studied, the excitation force should not exceed 100 N . In this research, especially resonance frequencies and mode shapes are of interest. In this context it is essential to know the impact of measurement variability on mode shapes.

MAC matrix in original situation									**MAC matrix in case 1 frf is changed**								
FEA EMA	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**FEA EMA**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
1	98.5	1.8	0.7	1.6	0.9	1.0	6.2	7.4	**1**	97.9	1.6	0.8	1.6	0.8	0.9	6.2	7.2
2	0.5	93.5	0.2	1.1	0.3	0.3	1.0	5.8	**2**	0.6	93.5	0.1	1.1	0.3	0.3	1.0	5.8
3	0.0	0.5	97.4	4.9	3.4	2.6	1.6	2.8	**3**	0.0	0.6	97.1	4.9	3.3	2.5	1.7	2.8
4	0.0	2.8	0.0	85.1	3.4	2.3	1.1	7.4	**4**	0.0	2.7	0.1	85.0	3.5	2.3	1.1	7.4
5	0.1	0.0	0.0	2.6	81.4	11.4	1.3	1.7	**5**	0.2	0.0	0.0	2.6	81.5	11.5	1.2	1.7
6	0.0	0.0	0.1	1.0	11.5	76.7	2.6	4.9	**6**	0.0	0.0	0.1	1.0	11.4	76.6	2.6	4.9
7	3.3	0.1	0.0	0.1	0.2	1.8	74.3	22.8	**7**	3.2	0.1	0.0	0.1	0.3	1.9	74.1	22.7
8	0.0	3.6	0.1	0.7	0.1	0.5	13.0	55.4	**8**	0.0	3.5	0.2	0.7	0.1	0.5	13.0	55.4

Table 4: comparison of two MAC matrices when 1 of 77 frfs is changed

The mode shapes from two measurements from table 3 are considered, namely measurement 1 and 9. Both measured frequency response functions considered are part of the same set of 77 measured frequency response functions. Both frf sets have one different frf, although for the same measurement point. Both experimental frf sets are translated to a set of eight mode shapes and each compared to a reference set of eight numerical mode shapes. Table 4 shows the two resulting MAC (Modal Assurance Criterion) [16] matrices. From table 4 it is evident that most MAC values change a little in case 1 of the measured frequency response functions is being altered. As is the case in table 3, the MAC value according to mode 1 varies the most and for mode 2 there is no change. The average relative change is little and only about 0.12 %. However, it has to be mentioned that in this case only one frf showed variability. If measurement variability of all frequency response functions would be taken into account, possibly more variability in MAC values would be experienced. At this stage no further research has been done in this area.

5 Model updating

In this section the updating of the numerical models is discussed. As mentioned in section 3 a so-called SVS numerical model is used for the sandwich panels. Some design parameter values were experimentally determined and others were purely estimated. In fact the design parameters that govern the model panel mass were updated in section 3. Elastic properties of the homogenised skin and core certainly need to be updated. In this research resonance frequencies and mode shapes of freely suspended panels are considered.

5.1 Global model updating

Table 5 shows the resonance frequency differences before any model updating was performed. The panel considered is a Monopan honeycomb panel with dimensions 595x420x15 mm. It is freely suspended.

#	FEA	f (Hz)	EMA	f (Hz)	Diff. (%)	MAC
1	1	74.941	1	83.750	-10.52	98.5
2	2	124.65	2	133.75	-6.80	93.7
3	3	194.43	3	210.00	-7.42	97.4
4	4	258.07	4	291.75	-11.54	84.5
5	5	277.25	5	325.00	-14.69	80.7
6	6	358.36	6	350.25	2.32	76.5
7	7	396.81	7	412.75	-3.86	74.9
8	8	407.07	8	438.75	-7.22	54.4

Table 5: resonance frequency differences and MAC values before model updating

From table 5 it is obvious that response differences before updating are rather high. As mentioned in sections 2 and 3 some skin and core elastic material constants have been estimated. However for some important properties as skin elastic moduli E_1 and E_2, core shear moduli G_{13} and G_{23}, initial values were experimentally determined (see section 3). Together with G_{12} of the skin these elastic properties are the parameters that most govern the panel’s resonance frequencies. Differences between EMA (experimental modal analysis) and FEA (finite element analysis) are due to erroneous design parameter values, restrictions of the FE model, measurement errors.

#	FEA	f (Hz)	EMA	f (Hz)	Diff. (%)	MAC
1	1	83.863	1	83.750	0.14	98.5
2	2	134.26	2	133.75	0.38	94.1
3	3	210.50	3	210.00	0.24	97.3
4	4	290.37	4	291.75	-0.47	84.4
5	5	301.85	5	325.00	-7.12	72.0
6	6	365.33	6	350.25	4.30	75.2
7	7	401.48	7	412.75	-2.73	74.7
8	8	435.66	8	438.75	-0.70	54.2

Table 6: resonance frequency differences and MAC values after model updating

Model updating is performed using Femtools 3.4, a specialist software package for combining finite element modal analysis and experimental modal analysis. Some restrictions were made to the model updating process. First of all some of the elastic properties must have nearly equal values, as is the case for the skin elastic moduli E_1 and E_2 because of the symmetry in weft and warp direction of the fibre reinforced skin. To attain this parameter relations have to be added to the model updating process. For this updating process these parameter changes are set to be equal. The same is done for the core shear moduli G_{13} and G_{23}. A second restriction is the maximum allowed change of a certain parameter during the updating process. For every parameter this is set in accordance to a predefined probability interval of the considered parameter.

From table 6 it is evident that the model updating process is most successful for mode pairs with a very good MAC value. Some of the higher modes are strongly coupled due to high damping (e.g. 3.1 %). Modal analysis was performed in LMS Test.Lab and frequency response function synthesis was used to overcome the problem of coupled modes. However, the most important reason for the frequency differences is the shortcomings of the FE model. In section 3 it was mentioned that the skin is modelled as an orthotropic material. In fact this is not a fully correct representation of the true skin as in reality the skin is a non balanced laminate. The asymmetry results in the existence of a membrane – bending coupling matrix B, together with matrices A, D and the transverse shear matrix S. For reasons of simplicity the orthotropic skin consideration is kept though as the overall FE model performance is very good.

In a further updating step the Poisson ratios of skin and core material are considered as updating parameters. Updating these parameters leads to a slight overall improvement of the different MAC values of approximately 0.5 % and a frequency difference improvement of about 1 %.

5.2 Local model updating

The objective of the global model updating described above in 5.1 is to improve the difference between measured and calculated modes. Model parameter changes are equal for all elements of the same kind, for example all skin elements. To study the effect of local parameter changes the inverse problem [17] can be solved; a local model updating can be performed. Here model parameters are updated on an element level and thus can vary from one element to another. Currently further research is done in this area.

6 Conclusions

In this work a study about thermoplastic honeycomb panels with fibre reinforced skin is presented. The objective of this work is to establish a sufficiently accurate FE model to predict panel natural frequencies and mode shapes. An important part of this work is the experimental determination of panel resonance behaviour. Special attention is paid to measurement uncertainty. The model updating procedure is described and illustrated. Model updating is performed to minimize the difference between calculated and measured resonance frequencies and mode shapes.

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